Machinery Messages

Case History

Identification of a unique vibration problem

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Alabama Power Corporation's Barry Steam Plant, located in Bucks, Alabama, operates a coal-fired, Westinghouse 270 MW cross compound steam turbine generator. This unit is composed of two separate machine trains - a high pressure (HP) train and an intermediate pressure (IP) train (Figure 1). The HP train consists of a high pressure turbine and a low pressure (LP) turbine driving a 130 MW generator. The IP train is made up of an IP turbine, an LP turbine and a 130 MW generator. The trains are cross connected with steam piping from the HP to the IP and to each LP. The turbine and generator rotors are supported by bearings at each end.

In March of 1988, the generator rotor of the IP train was removed and sent out for a complete rewind and high speed thermal balance. The rewind was accomplished using the existing copper bars. The retaining rings were thoroughly inspected, and the rotor was high speed balanced at operating temperature. The unit was back in operation on 20 June 1988.

Shortly after being placed in operation, a vibration problem was noticed primarily on bearings #4, #5, and #6 of the IP train. Monitor and strip chart readings indicated both increasing and decreasing step changes in the vibration amplitudes occurring intermittently from 30 minutes to two hours after the unit was loaded. The amplitude of the vibration step changes ranged from less than 1 mil to over 6 mils (152 μm) peak-to-peak, varying with each change.

Corrective attempts

Initial actions taken by maintenance personnel to diagnose and correct the problem had little effect. Both the LP turbine and the generator seal clearances were checked and found to be within tolerances, but nevertheless, were changed. A check of the alignment at the generator and LP turbine coupling found nothing unusual. Bore inspection of both ends of the generator under the retaining rings indicated nothing unusual as did a check of the bearing clearances on the generator rotor. Increasing the loading of the bearings also failed to correct the problem. Impedance and recurrent surge testing on the generator, performed in the hope of identifying shorted generator turns, revealed nothing. A known loose balance weight in the LP turbine was corrected and still there was no effect on the step changes.

Living with the problem

While experimenting with various operating modes during a vibration step change, it was found that, following a large amplitude-increasing change, the vibration amplitudes would decrease upon restart when the unit was shut down to turning gear and immediately restarted. During the coastdowns, step changes (both increasing and decreasing) were found to take place around 1800 rpm. However, the startups to 3600 were always smooth. Minor and mode-
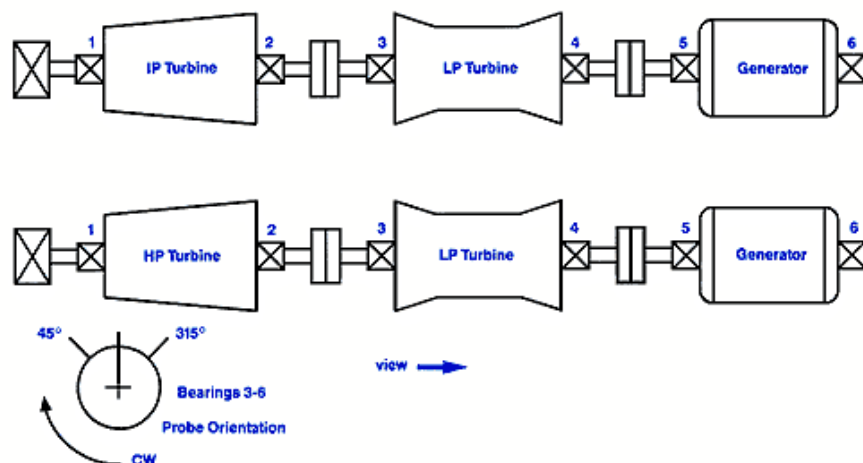

Figure 1
Barry Steam Plant's Westinghouse 270 MW cross compound steam turbine generator unit.
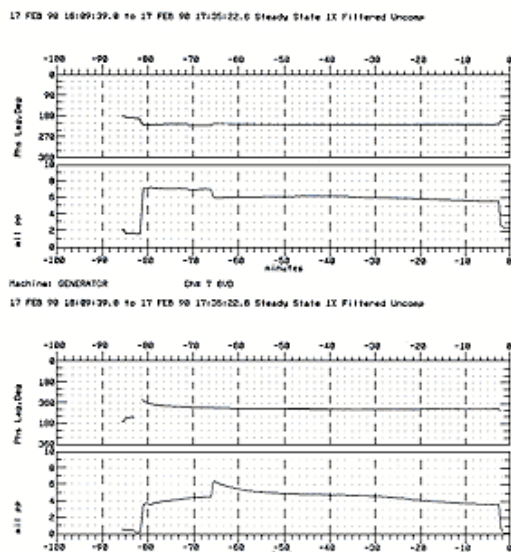
Figure 2
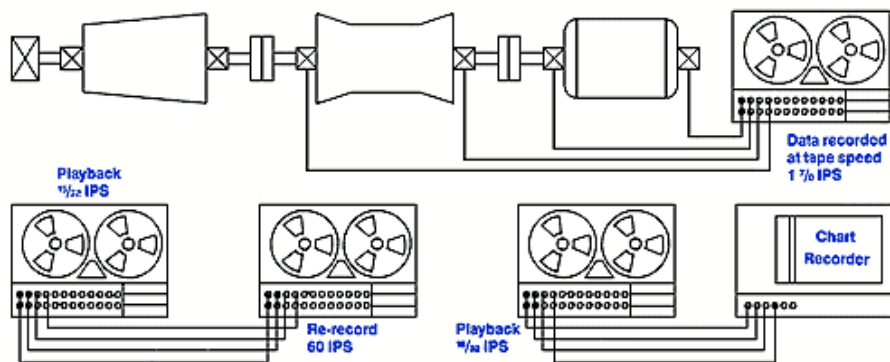Data showing increasing step change in vibration amplitude at bearing #5 and #6.

Figure 3
Process used to slow the event for analysis.

on bearings #3 through #6 (Figure 1), and temporary field wiring was routed to the control room where an instrumentation-grade data recorder was set up. The plant rented a fourteen-channel data recorder and began 24-hour taping of the transducer signals.

The first tapes that arrived at Bently Nevada on 18 February 1990 showed a rather severe 6 mils (152 μm) pp increasing step change in the vibration amplitude on the vertical transducers at bearings #5 and #6 as shown in Figure 2. The horizontal transducer at bearing #5 indicated a somewhat lower 3.5 mils (89 μm) pp change, while the horizontal readings at bearing #6 were still lower at 2 mils (51 μm) pp. Bearings #3 and #4 had not exceeded 2 mils pp (51 μm) in amplitude change.

Localizing the problem

It was believed that the higher amplitudes at bearings #5 and #6 indicated the problem was originating in the generator and affecting the LP turbine at bearings #3 and #4. The vibration signals were originally recorded on an FM analog data recorder at a tape speed of 1-7/8 in/sec. For a machine running at 3600 rpm, this meant that at normal play back speed, one revolution of the shaft took 0.0166 seconds, too fast to distinguish an event happening in less than one revolution.

Figure 3 illustrates the process used to slow the event down enough to be analyzed. The data was played back at 15/32 in/sec, and re-recorded on a second tape recorder at 60 in/sec. This new tape was then played back at 15/32 in/sec (512 times slower than the original speed) to an analog chart recorder and a camera-equipped oscilloscope. One revolution of the shaft now took 8.533 seconds — long enough for an accurate documentation to indicate which signal was changing first.

Figure 4, the direct timebase trace of bearings #4 and #5 vertical transducers during a vibration step change, shows a 1-3 millisecond lead in the signal change at bearing #5, indicating the source of the problem to be in the generator as ▶

rate step changes occurring during severe load demand periods (when the unit could not be cycled) were minimized by limiting the megavar loading (i.e., limiting the field temperatures). Additionally, the vibration amplitudes at some locations would settle over a period of time even at a constant megawatt and megavar loading.

As several months passed and the step changes still occurred unpredictably, the frequency of the step changes increased to an almost daily occurrence. In February 1990, the Machinery Diagnostic Services (MDS) group of Bently Nevada was commissioned to document and analyze the step change data and provide an opinion on the cause and corrective action needed.

Gathering the data

It was decided that, because of the unpredictable nature of the load demand, and consequently the unit operation and the step changes themselves, the most efficient and cost-effective method to acquire data would be for plant personnel to tape-record the vibration signals and then send the data tapes to Bently Nevada for diagnostic analysis.

The unit had originally been instrumented with shaft riders connected to monitors in the control room. However, to acquire more accurate vibration data, proximity transducers were required.

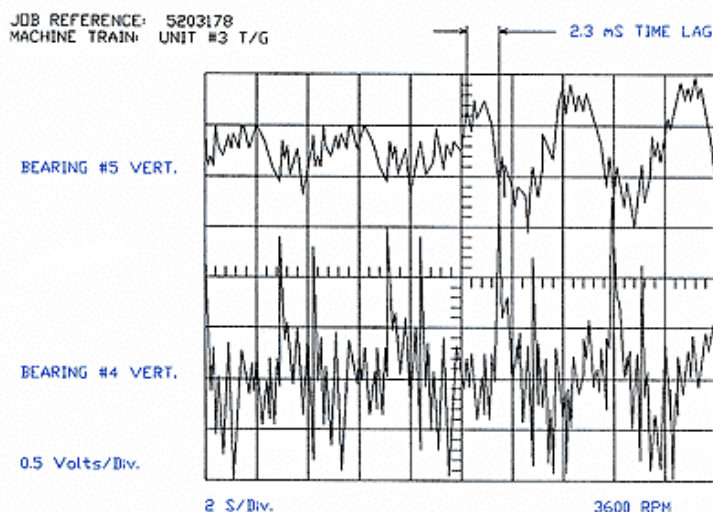Bently Nevada proximity transducers were installed in an XY configuration

Figure 4
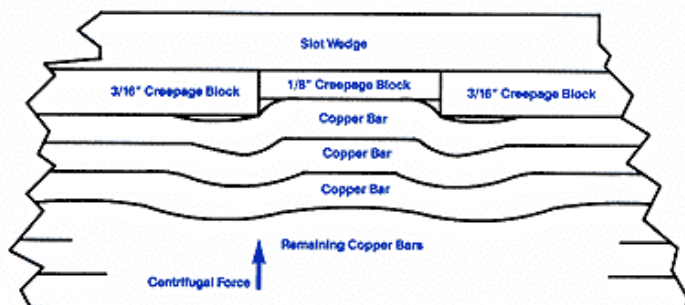Direct timebase of bearings #4 and #5 vertical transducers.

Figure 5
(Exaggerated) example of copper bar deformation due to centrifugal force and non-uniform blocking.

suspected. A comparison of the step changes from bearings #5 and #6 at each end of the generator did not reveal any discernable time difference in the signal change.

Theories

As the step changes continued, several causes were suggested. Reasoning quickly eliminated most of them. The speed and the intermittency of the amplitude changes ruled out the possibility of foundation, shorted turn, cracked shaft, or misalignment problems. The harmonics usually present in a rubbing problem were not present, and a rub should have rubbed out long ago. The source of the perturbation had already been shown to take place in the generator first, so the rest of the turbine

could be eliminated as the cause. Subsynchronous frequency components, typical of a whirl or whip problem, were not present, eliminating this possibility. Any kind of slipping between the LP turbine and generator coupling could not account for amplitude increases and decreases, and the changes in the LP rotor always followed the generator rotor in time.

Loose blocking in the generator rotor was briefly considered, but ruled out since starting the unit immediately after a shutdown from a high vibration step change did not seem to produce any shift in the blocking. The heat load in the rotor should have changed little during the brief coastdown period, and the freedom of the blocks to respond to the changes in centrifugal force should have

still been high. The rotor should have increased to the same vibration amplitude seen just prior to being shut down, but it never did. The other theory involving loose blocking held that one or more loose blocks had become trapped between the end face of the coil end windings and the inner face of the end retaining ring. This would in turn prevent proper axial growth of the copper and cause a rotor bending force variable with generator load and/or excitation. This scenario did not fit the non-load dependency and the speed of the step change.

One of the other theories was loose retaining rings, especially as this particular problem has been known to cause sudden catastrophic rotor failure. This hypothesis was eventually eliminated because the characteristic symptoms of this problem did not fit the step change data.

Problems with the generator wedges were also among the most likely possibilities. During the rewind of the rotor, eleven new wedges were manufactured through reverse engineering. The flaw in this theory was that the step changes seemed to be more load dependent than speed dependent. If the wedges were loose in their slots, shifts in rotor vibration due to a radial shift in the wedges and copper should have been fairly constant and predictable during shutdowns and restarts once the unit was up to operating temperature. They were not.

The working theory

Copper windings sticking in the slots became the most likely candidate that would explain all of the machine symptoms in this two pole generator rotor. The scenario is as follows: improperly sized wedges or blocking insulation beneath the wedges might be binding the copper bars when the rotor was at speed. This could also prevent the windings from expanding axially when heated. Under normal circumstances, this would have resulted in a slowly developing rotor bow, yet if this were

taking place in a majority of slots on both poles (both sides of the rotor), the binding forces would cancel themselves out, building up axial strain forces within the rotor.

If one or more of the windings on predominately one side of the rotor broke free, allowing the rotor to bow instantly, it would cause the vibration step change. The amount of step change would depend on how much thermal strain was in the copper, which would be a function of unit watt and var load (field temperature), the number of windings released suddenly, and their angular location. In addition, the direction of the bow could either add to or subtract from any residual imbalance already present in the rotor, causing either upward or downward movements in the overall vibration amplitude. Adding to the unpredictable nature of the step changes was the fact that the more the unit bound and released, the more the materials binding the windings would wear. In theory, this would both dust and lubricate the windings, and increase the sliding clearances, thus improving the situation.

In September 1990, the plant took the unit off line for an inspection and overhaul. The generator rotor was returned to the repair shop that did the original rewinding and carefully disassembled. As suspected, it was found that the G-11 type insulation (creepage blocking) beneath the slot wedges had been improperly sized. Varying sizes of blocking had been installed during reinstallation of the slot wedges.

Figure 5 illustrates, in an exaggerated form, how the copper bars deformed because of centrifugal force and nonuniform blocking, preventing them from expanding axially. The final repair resolution involved slightly milling the underside of the slot wedges in order to use a uniform strip of creepage block. This created a smooth slip plane between the copper and the blocking insulation, allowing uniform axial expansion of the copper bars.

The unit was back in service 26 November 1990, and has not experienced any further vibration step changes.

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Case History

The importance of monitoring Shaft Centerline data

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This article discusses the basic theory behind Shaft Centerline monitoring and the problems experienced in a recent field study where startup assistance was provided on main process machinery. A case history is also included which focuses on severe misalignment problems experienced during initial machinery startup. The problem was further compounded by a loose bearing retaining setscrew. This case history demonstrates that Shaft Centerline information is an important part of the diagnostic process as it provides additional information on complex machinery malfunctions.

Introduction

1X and/or 2X Polar and/or Bode plots contribute to being the most useful machinery diagnostic data. Both plots show a machine's amplitude and phase response versus speed. In practice, this type of data presentation is widespread and has become accepted as an industry standard as the "best" plots to observe for machinery diagnostics. However, another data presentation type, Shaft Centerline, is generally overlooked.

Many machinery diagnosticians have long recognized the value of shaft radial position information. The existence of a preload (external or internal) can often be quickly identified by observing the average radial position of the Shaft Centerline within the bearing clearance. Shaft Centerline position may also change as a result of bearing babbitt deterioration due to electrostatic discharge or simply as a result of bearing wear resulting over time.

Monitoring Shaft Centerline data in today's rotating machinery applications can provide important and relevant information to changing machinery conditions.

This concept is illustrated in the following case history in which Shaft Centerline data became the key component in solving a machinery malfunction. But first, a brief discussion is in order to clarify some theoretical aspects regarding Shaft Centerline monitoring and diagnostics.

Shaft Centerline theory

When a noncontacting eddy current proximity transducer and Proximitor® are used to monitor lateral shaft motion, the transducer system provides the following signal components:

1. An AC signal (in this case, negatively fluctuating) which provides shaft dynamic motion relative to the probe mounting.

2. A DC signal which provides the average radial shaft position relative to the probe mounting.

Typically, the dynamic signal is monitored by a radial vibration monitor and is displayed as the amount of overall machine vibration in mils or micrometres peak-to-peak. However, the DC component of this signal is, for the most part, unused. Therefore, it is relevant to note that, just as an axial position probe monitors axial movement, so can a radial proximity probe be used to measure radial shaft position within the bearing.

A typical arrangement for Shaft Centerline monitoring is to have two orthogonal (XY) proximity probes per ▶